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DESIGN OF A FATIGUE-RESISTANT PRESS USED TO CONTAIN
3,000 KIPS OF END LOAD AS A RESULT OF HIGH PRESSURE
CYCLIC TESTING OF OPEN-ENDED CYLINDERS

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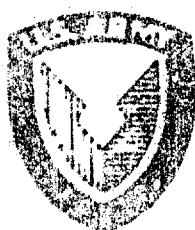
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To determine the safe service life of cannon tubes, several samples must be cyclically tested to failure. Since testing these large diameter, open-ended cylinders requires pressures as high as 100 ksi, the fatigue-loading conditions on the testing equipment are very demanding. One method for reacting the very high-end loads associated with testing these open-ended cylinders is through the use of an external press. The press design discussed in this report deals with the design procedures involved in developing an infinite life press that can react end loads as high as 3000 kips. This press design involves the use of two low deflection platens connected by two high strength posts. To design an infinite life threaded post, several methods of fatigue life enhancement can be applied. The most effective method of life extension is to preload the threaded post producing a high mean stress, but reducing the alternating stress or stress amplitude. This avenue of enhancement became the method of choice. Producing sufficient preload on a post with a diameter of 7 inches and a length of 115 inches by mechanical means requires torque-producing equipment presently available. By using electric heating elements inserted at the center of the posts, the posts can be expanded and the preload applied.					
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INTRODUCTION

When designing cannon systems, the durability (fatigue) life of the cannon tube must be measured (ref 1). Requirements state that the fatigue test must be performed in the open-ended condition. Since cannon tubes have relatively large bore diameters and are tested at pressures as high as 100 ksi, significant end loads are produced. Historically, two methods of reacting these end loads have been used: a large capacity press (Figure 1) or a mandrel (Figure 2).

The mandrel method utilizes a large solid steel bar that passes through the center of the test specimen. Sealing closures are then slid over each end of the mandrel and into the gap between the mandrel and the specimen. The ends of the mandrel are threaded so that large nuts can secure the sealing closures in place. High pressure fluid is pumped through a small, angled porthole in the mandrel. The fluid enters the test specimen from the mandrel at a point between the two sealing closures. With this scheme, the only end loads reacted by the mandrel are those generated by the pressure acting on the sealing area between the mandrel and the test specimen. This relatively small axial load is combined with external compressive loads as a result of the pressurized fluid between the test specimen and the mandrel. These combined loads coupled with stress concentration factors associated with the fluid exit port result in a very short mandrel fatigue life.

The press method allows the end loads to be reacted from outside of the test specimen. The magnitude of this end load is quite high since the test pressure acts on the entire cross-sectional area of the sealing closure. These end loads can be as high as 3000 kips. It is difficult to design and build a compact press that will survive extended cyclic loading at these loading levels. Increasing the fatigue strength of the press is possible by preloading the posts in a manner similar to preloading a bolt. In this report, we will demonstrate the design method used to develop a compact, high capacity, fatigue-resistant press rated for continuous service at 3000 kips. Of particular interest, is the innovative method used to preload the press posts.

DESIGN METHODOLOGY

The objective was to design a press with a life of 1,000,000 cycles at a maximum externally applied load of 3000 kips with a minimum safety factor of 2.0. The material to be used was restricted to the available materials for cost and convenience considerations. The top and bottom platens from existing non-preloaded presses were to be utilized. The press was to accommodate test specimens as long as 65 inches. Previous studies (ref 2) have shown that the likely fatigue failure location on presses used to react end loads is the press post. The thrust of this study was to design press posts that will survive the design requirements stated above. To meet these requirements, a design similar to the design shown in Figure 1 was used. Namely, there is a top and bottom platen connected by two posts. Since the new press was to be preloaded, it became evident that platen adjustability, with regard to specimen size, would have to be omitted. Once again, the bulk of the design was to determine the conditions under which two posts of approximately 7-inch diameter subjected to cyclic loading ranging from zero to 3000 kips could survive 1,000,000 cycles.

BASIC CONCEPT

The platens of the preloaded press are held in place by threaded nuts. The mating threads on the posts produce stress concentration factors that are the source of fatigue crack initiation and failure. To design a fastener connection that eliminates this stress concentration is beyond the scope of this work. The preloading of bolts has been a long-standing technique in extending the life of a threaded connection (ref 3). This method can be applied to the press with some small modifications. By placing a large washer or compression sleeve around the posts between the two platens, each post can be preloaded to a specific value. This preload reduces the damaging effects of large alternating stresses, S_A (also known as stress amplitude). Figures 3a, 3b, and 3c illustrate the advantages of reducing the alternating stresses. A preloaded press arrangement is shown in Figure 4.

The materials used to construct the structural components of the press are described with their mechanical properties in Table 1. Each platen is 17 inches high, 16 inches deep, and 48 inches wide. Two holes large enough for the press posts to pass through are machined symmetrically at each end. The compression sleeves are 65 inches long with an inner diameter (ID) of 8 inches and an outer diameter (OD) of 12 inches. The nuts have an ID of 7 inches and an OD of 13 inches. The posts are 115 inches long with an OD of 7 inches. The posts were threaded approximately 11 inches from each end. The threads used were 7-4UNRC-2A, with 0.027 inch minimum root radius. After machining and heat treating, the threads on the posts and the nuts were shot peened with 0.008 diameter shot to 0.012A intensity and 100 percent coverage. Other stress-reducing features were applied to the post and the nut. Each end of the posts received slight undercuts before the first thread, and each nut had a tapered nose (Figure 5). These two features help reduce the stresses at the first loaded thread (ref 3).

It is important to note that the use of marage material is not intended to imply that such material be incorporated into the ASME Pressure Vessel and Piping Code for High Pressure Vessels (Sect. VIII, Div. 3). The use of this material was an attempt to utilize a high-strength material with a high endurance limit, while minimizing the size of the components. The authors are aware of numerous problems associated with the use of this material in similar applications, including notch sensitivity and susceptibility to stress corrosion cracking. Its toughness limitations were also considered. We have addressed these problems by incorporating several safeguards into the press. Shot peening the threads will reduce the possibility of stress corrosion cracking by placing those areas most likely to initiate stress corrosion cracks in a state of compressive residual stress. Each post and sleeve was instrumented at the time of construction and continues to be monitored for residual stress. Should loss of residual stress be observed (indicating the presence of cracks), use of the press will be halted. A nondestructive inspection procedure applied at regular intervals is in effect. Once the presence of cracks is noted, the press will be removed from service. We anticipate that the press will survive its design life, but consider the incorporation of these safeguards as prudent.

FATIGUE LIFE PREDICTION METHOD

With preloaded connections, the tensile preloaded component is loaded such that it has a tensile minimum stress, and the compressively loaded part has a compressive minimum stress. When the cyclic load is applied, the compressive residual stresses in the compressively loaded part must be overcome before substantially increasing the tensile stress on the component with tensile residual stress. Therefore, the effect of the preload is to increase the mean stress on the tensile loaded component and reduce its stress amplitude during cycling. Many methods of analysis have been developed for determining the effect of tensile mean stress on the fatigue performance of components. The most commonly applied method is the Goodman approach, although success has been demonstrated by using the Gerber and Soderberg approaches (ref 4).

The most conservative of these methods is the Soderberg approach. With this method, it is assumed that the locus of points, which describes those combinations of mean stress, S_m , and stress amplitude, S_a , resulting in the same fatigue life, is a straight line on a plot of stress amplitude on the ordinate and mean stress on the abscissa. For a life of 1,000,000 cycles, the Soderberg line intersects the alternating stress axis at the endurance limit of the material, S_e , and intersects the mean stress axis at the yield strength, S_y , of the material. Combinations of mean stress and alternating stress that fall to the right and above the line will have lives shorter than 1,000,000 cycles, while combinations of mean stress and alternating stress left and below the line will have lives longer than 1,000,000 cycles. Figure 6 shows the Soderberg line for a material with an endurance limit of 100 ksi and a yield strength of 237 ksi. Also plotted in the figure are a short-life condition (above the line) and a long-life condition (below the line).

When determining the fatigue life of components that possess stress concentrations, the Soderberg line must be slightly altered. Instead of drawing the Soderberg line from the yield point on the mean stress axis to the endurance limit on the alternating stress axis, the line is drawn from the yield point on the mean stress axis to a value representing the endurance limit divided by the stress concentration factor on the alternating stress axis (Figure 7). The stress concentration factor, K_c , or in this case, the fatigue strength reduction factor due to cut threads, is commonly accepted as 3.8 (ref 5).

The design requirement of a minimum safety factor of 2.0 must be accounted for. This requirement is met when the plot of mean stress and stress amplitude falls below the line that intersects the ordinate at the endurance limit divided by twice the fatigue strength reduction factor and the abscissa at half the yield strength (Figure 8). By using a known formula for the safety factor as applied to the Soderberg criteria, these intersection points can be calculated for FS=2 as (ref 4)

$$S_y/FS = S_M + K_c S_A S_y/S_u$$

We have now defined the area within which any combination of mean stress and stress amplitude will yield a fatigue life equal to or greater than 1,000,000 cycles at a load of 3000 kips and with a minimum safety factor of 2.0.

FATIGUE LIFE CALCULATIONS

To determine the life of the press posts in a non-preloaded and a preloaded press, the stresses produced during operation must be determined. The stresses in the posts are the normal tensile stresses necessary to react the end loads, and the bending loads developed to accommodate the deformation of the platens. The normal tensile stress produced is the end load produced during testing divided by the area of the two press posts. The bending stresses are calculated as follows.

The platen is loaded as a beam with the total end load acting at its mid-span and supported by the two posts. Assuming the platen to be a simply-supported beam, the angle through which the post reaction points must rotate is given by the equation (ref 6)

$$\Theta = \frac{Wl^2}{48EI}$$

where Θ is the angle of rotation, W is the total load (3000 kips), l is the support span, E is the modulus of elasticity (29,000 ksi), and I is the moment of inertia. The platen is a solid rectangular block 16 inches deep and 17 inches high with a support span between the two press posts of 32 inches. To be conservative, the moment of inertia was calculated at the cross section where the posts pass through the platen. Using well-known formulae (ref 7), the value of I is

$$I = \frac{bh^3}{12} = 3275 \text{ in.}^4$$

Substituting these values into the above equation, we find

$$\Theta = 6.74 \times 10^{-4} \text{ radians}$$

Both platens rotate through this angle at the points where the posts pass through them. This bends the posts through the same angle. Since this is a pure bending load, the resulting radius of curvature of the post is (ref 6)

$$R \approx \frac{L}{2\theta}$$

where L is the length of the post and 2θ is the double angle produced when both platens deflect under load. The effective length of the press post for this calculation is the distance between the platens (65 inches). Substituting these values, we find that the radius of curvature of each press post is 48,234 inches. From basic bending theory (ref 8), the bending stress, S_B , produced by this deformation is

$$S_B = \frac{ED}{2R}$$

where D is the diameter of the press post (7 inches). Substituting into this equation, we find that the bending stress produced is 2.1 Kpsi.

Returning to the calculations for normal stress, the tensile stress area (ref 9) of a 7.00-4UNR-2A thread is 35.7 in.². From this value we must subtract 0.44 in.² because there is a 0.75-in. diameter hole in the center of each press post. The reason for these holes will be explained later with the preloading procedure. The resulting tensile stress area is 35.26 in.². Assuming symmetrical loading between the two posts, each post supports 1500 kips, producing a normal stress of 42.54 Kpsi. The total maximum stress produced in each post is the superposition of the normal and bending stresses or 44.64 Kpsi. The mean stress is then equal to the average of the maximum and minimum stresses or 22.32 Kpsi. The stress amplitude is also 22.32 Kpsi (see Figure 3a). This point is plotted in Figure 9 and clearly falls to the right and above the Soderberg line representing a safety factor of 2.0. The calculated safety factor for this combination of mean stress and alternating stress is 1.06. Therefore, the use of a threaded press post without preloading will result in a fatigue life that does not meet the safety requirements stated in the original objective.

When determining the appropriate amount of preload, an amount must be used to prevent separation of any of the components in the load train. Should the compression sleeve lose all of its compressive loading, the benefits of preloading the post will be lost. The minimum preload to prevent loss of compression in the compression sleeve is (ref 10)

$$F_i = (k_c/(k_p + k_c)) \times P_{MAX}$$

$$\frac{1}{k_c} = \frac{1}{k_s} + \frac{1}{k_p}$$

where k_s is the stiffness of the compression sleeve (28,032 kips/in.), k_p is the stiffness of the platen (177,000 kips/in.), k_c is the combined stiffness of the parts under compression (24,199 kips/in.), k_p is the stiffness of the posts (10,458 kips/in.), and P_{MAX} is the maximum externally applied load to each post (1500 kips). Substituting, we find that the minimum preload is $F_i = 1047$ kips.

This is the theoretical minimum preload value. To insure that there is no loss of compression of the compression sleeve, a preload of 1200 kips was used. Now we must use this value to determine the cyclic forces and stresses in the preloaded post using the following formulae. The maximum force applied to the post is (ref 4)

$$F_{MAX} = (k_p/(k_p + k_c)) \times P_{MAX} + F_i$$

$$F_{MAX} = 1652 \text{ kips}$$

With F_i and F_{MAX} as the minimum and maximum applied post-loads during the fatigue cycle, we can determine the mean load and load amplitude and subsequently, the mean stress and stress amplitude. Using the tensile stress area and bending stress found earlier, the mean stress on the preloaded post is 40.8 ksi and the stress amplitude on the preloaded post is 6.76 ksi. This point is plotted in Figure 9 and clearly falls to the left and below the Soderberg line for $FS=2$. The calculated safety factor is 2.33.

We are now assured to have a life in excess of 1,000,000 cycles. Not included in this quantitative analysis are the benefits from shot peening, as well as the other stress-reducing geometric features listed earlier.

APPLICATION OF PRELOAD

The necessary preload was applied in the following manner. The total elongation of the press post to achieve a preload of 1,200,000 pounds equals the elongation of the post plus the compression of the sleeve and the compression of both pistens

$$\delta_t = \delta_p + \delta_s + 2\delta_{pl}$$

where $\delta = F_t L/AE$; L = length of component in question; A is the area of the component in question; and E is the elastic modulus of the component in question. Substituting, we have $\delta_p = 0.1097$ in., $\delta_s = 0.0428$ in., $\delta_{pl} = 0.0068$ in., and $\delta_t = 0.1661$ in.

To produce a deflection of 0.1661 in., a nut of pitch 0.250 in. must be turned 0.1661/0.250 or 0.664 revolution or 239 degrees. The torques required to produce this deflection are beyond the means of production at our facility. To overcome this shortfall, the following procedure was used.

The press was assembled and the nuts were mechanically torqued to 600 ft-lbs to assure a uniform initial condition. Four heating elements, approximately 0.750 inch in diameter and 36 inches long, were placed in holes drilled down the center of each press post (Figure 10). One heating element was inserted in each end of each post. The heating elements provided a total of 22,000 watts of power and a watt density of 78 watts/in.². Sufficient heat was applied to thermally expand the posts, allowing 239 degrees of nut rotation. Once this nut rotation was achieved, the press was allowed to cool. The sleeves prevented the posts from returning to their original length, thus applying the preload. Monitoring methods were used to ensure that the preload was actually applied. The sleeve and posts were instrumented at the time of assembly. Strain gages were applied and residual stresses were measured. The results of these measurements suggested that sufficient preload had been applied.

The minimum increase in post-temperature required to produce the preload is calculated as (ref 11)

$$\Delta T = \delta_t / (L\alpha)$$

where α is the linear coefficient of thermal expansion or 5.82 E-6 in./F at 500° F. Substituting, we find that the temperature increase necessary is 293° F. This temperature rise will not result in any adverse effect on any material component used to construct the press.

CONCLUSIONS

It has been shown that inducing a preload in the threaded posts of a high capacity press significantly increases the life of the connection. The application of this preload will alleviate the failures of test fixturing associated with the testing of open-ended cylinders subjected to high pressure cyclic loading. The press was constructed utilizing high-strength materials for convenience. Although there are potential problems associated with the use of such materials, when appropriate care is taken in monitoring the performance of the assembled press, we are convinced that long life and safe operation is possible with this construction method.

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MATERIAL PROPERTIES

COMPONENT	MATERIAL	.2% YIELD STR (ksi)	ENDURANCE LIMIT (ksi)
Platens	AISI 4140	80 (min)	42
Sleeves	A519	95 (min)	62
Nuts	A723	140 (min)	72
Posts	Marage Stl Grade C250	237	100

Table 1

2-POST NON-PRELOADED PRESS

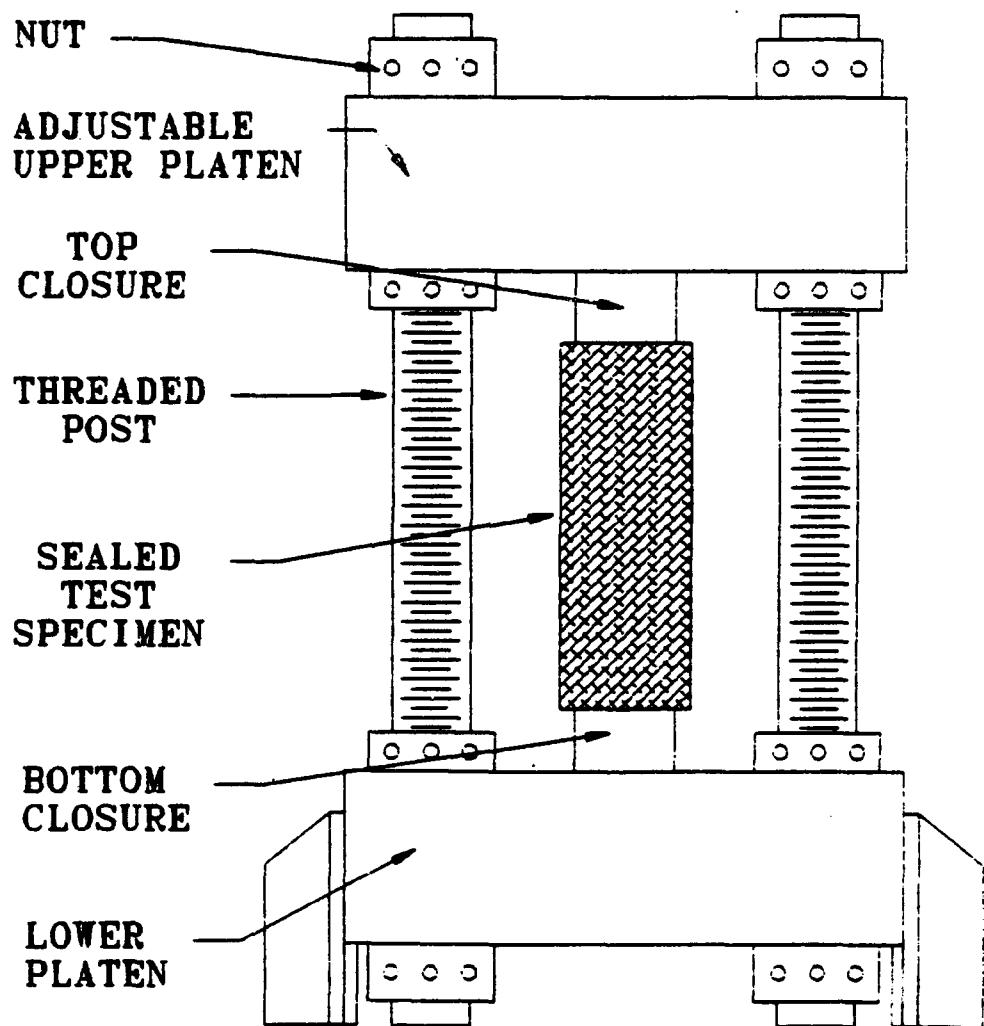


Figure 1

HIGH STRENGTH TESTING MANDREL

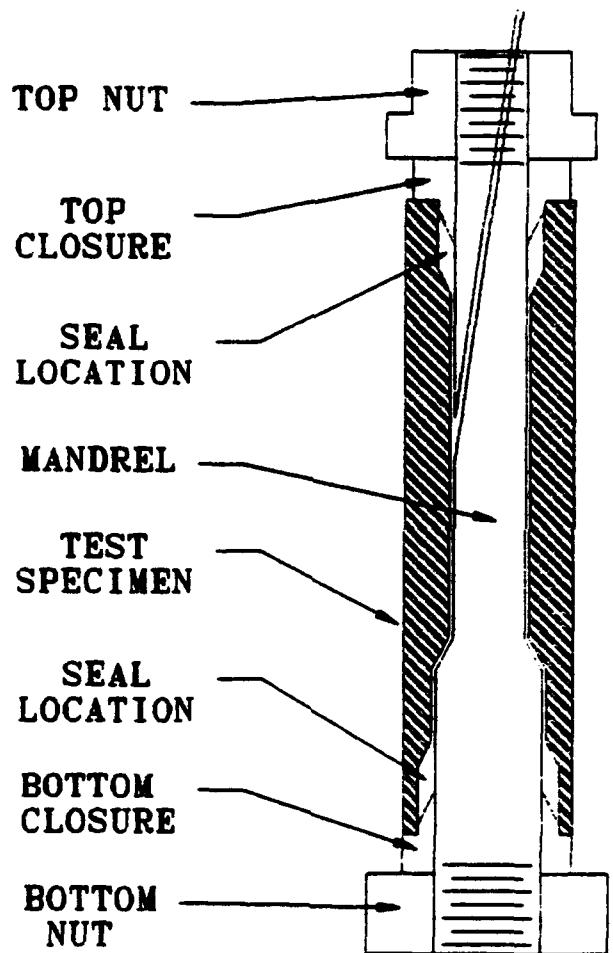


Figure 2

STRESS AMPLITUDE vs TIME NO PRELOAD

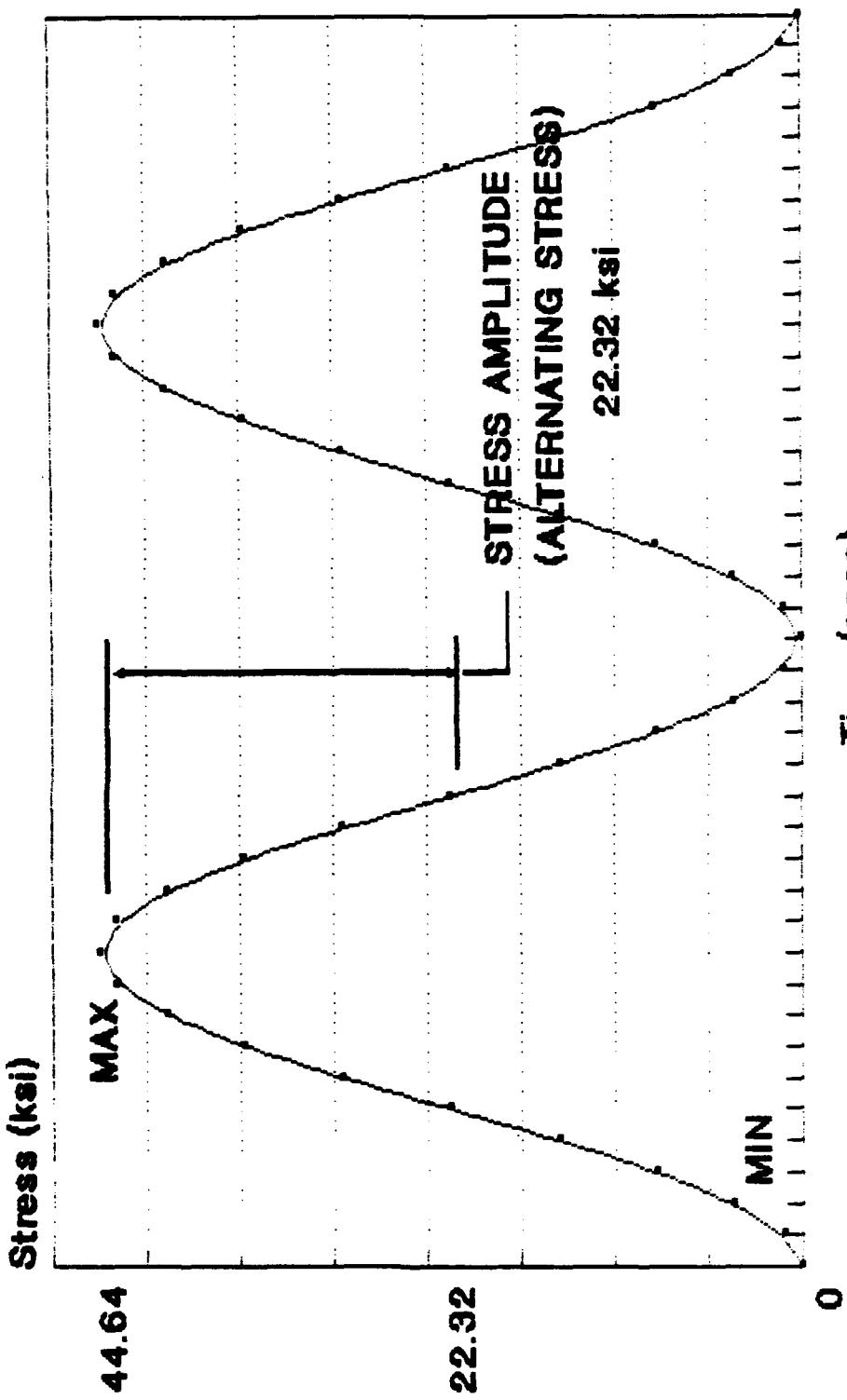


Figure 3A

STRESS AMPLITUDE VS TIME PRELOADED

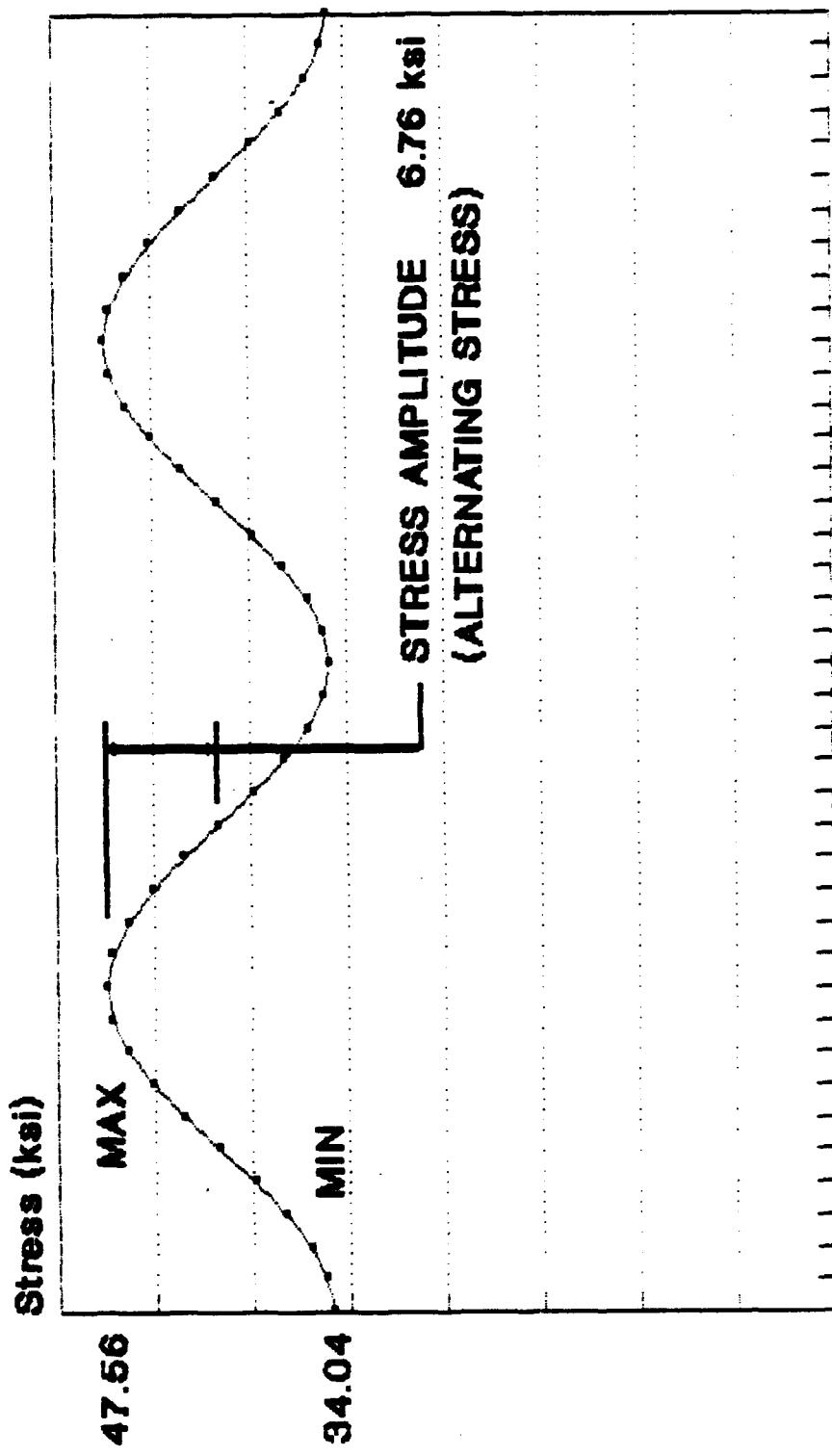
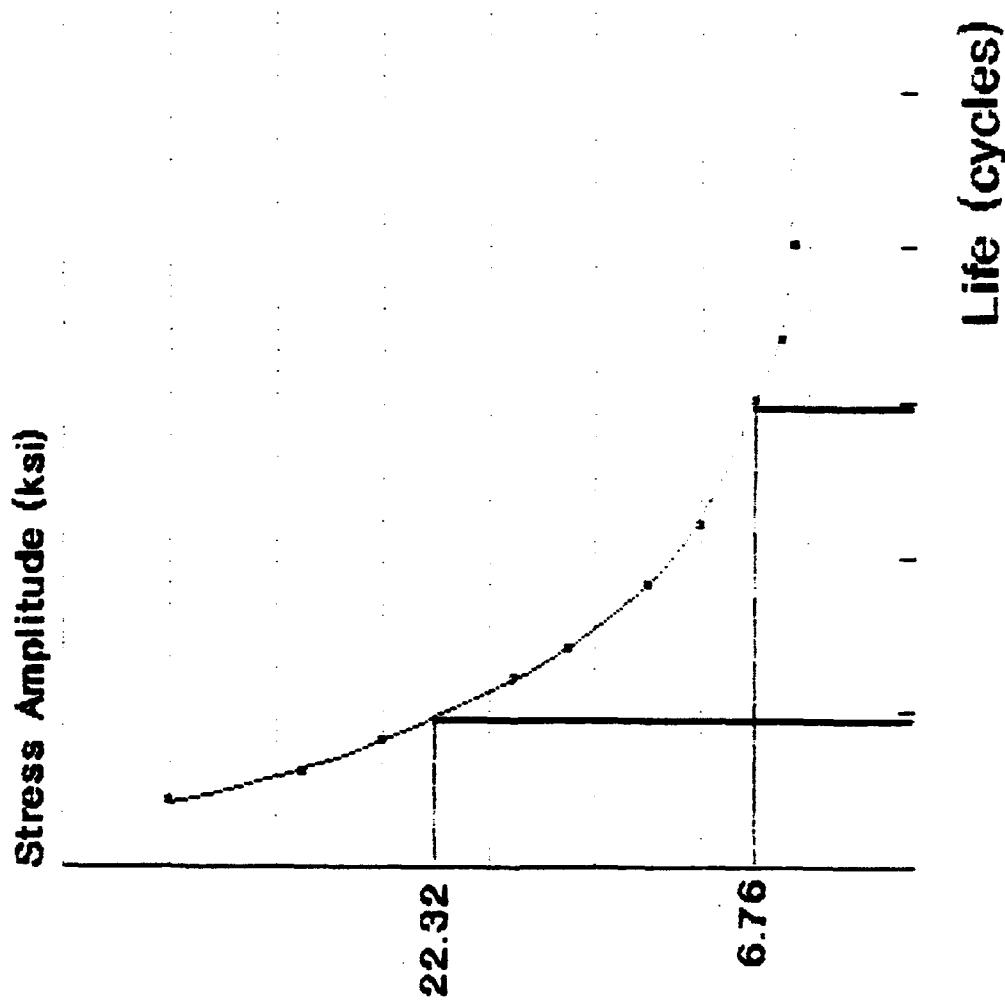


Figure 3B

STRESS AMPLITUDE VS LIFE



2-POST PRELOADED PRESS

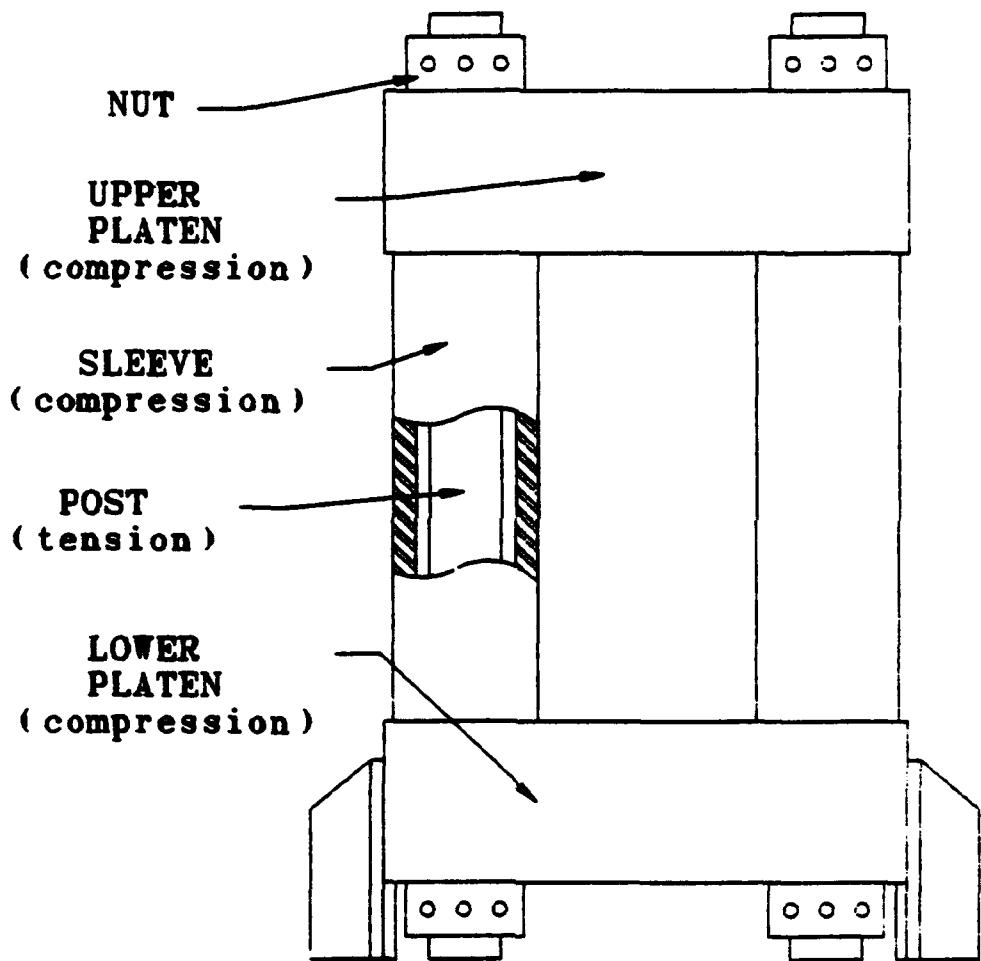


Figure 4

STRESS REDUCERS

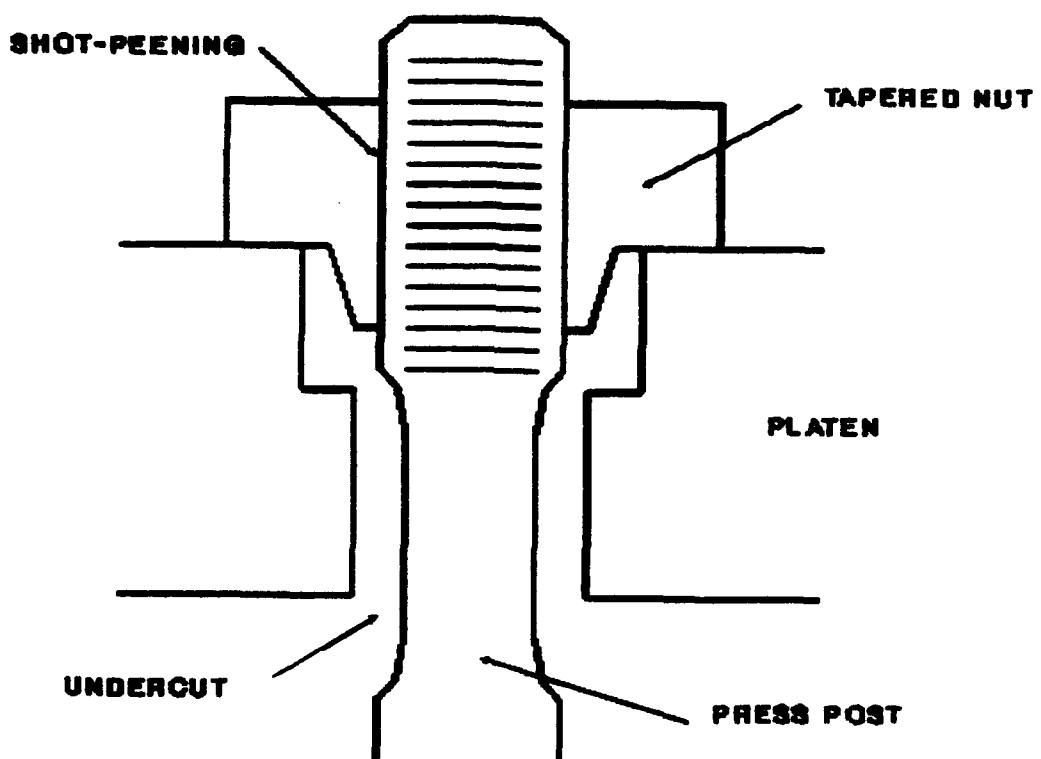


Figure 5

FATIGUE LIFE PREDICTION

SODERBERG CRITERIA

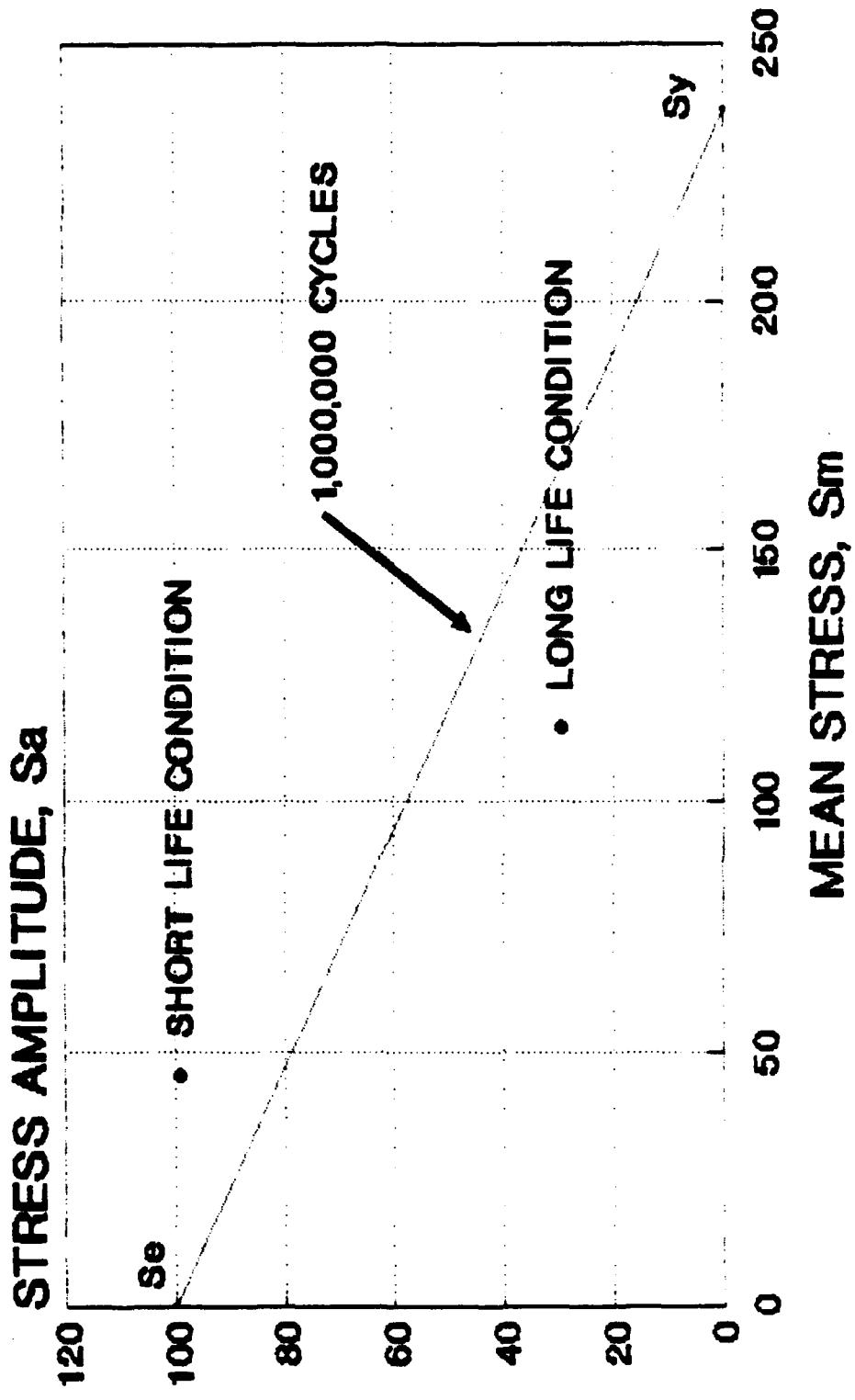


Figure 6

SODERBERG CRITERIA ADJUSTING FOR STRESS CONCENTRATIONS

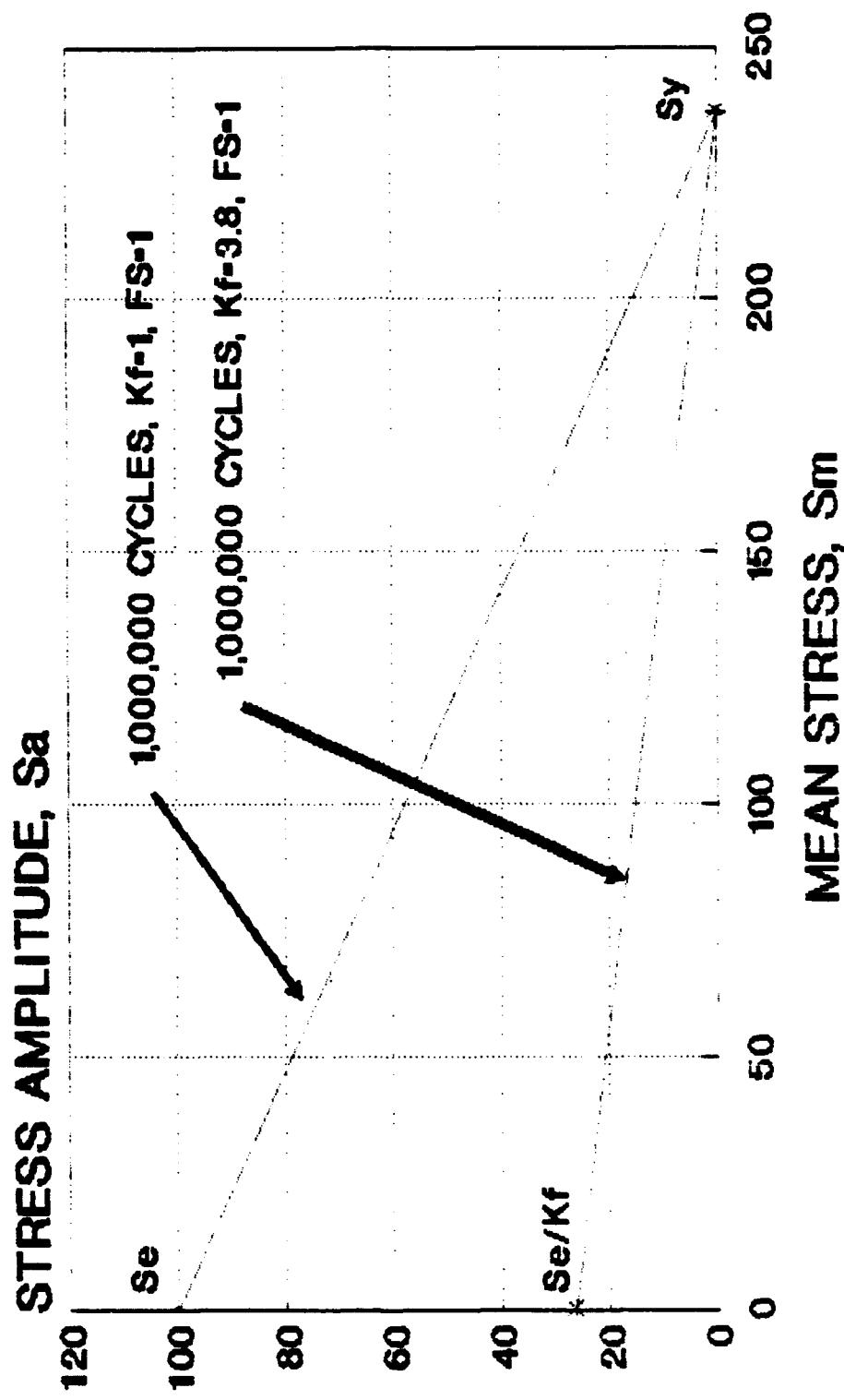


Figure 7

SODERBERG CRITERIA ADJUSTING FOR FACTOR OF SAFETY

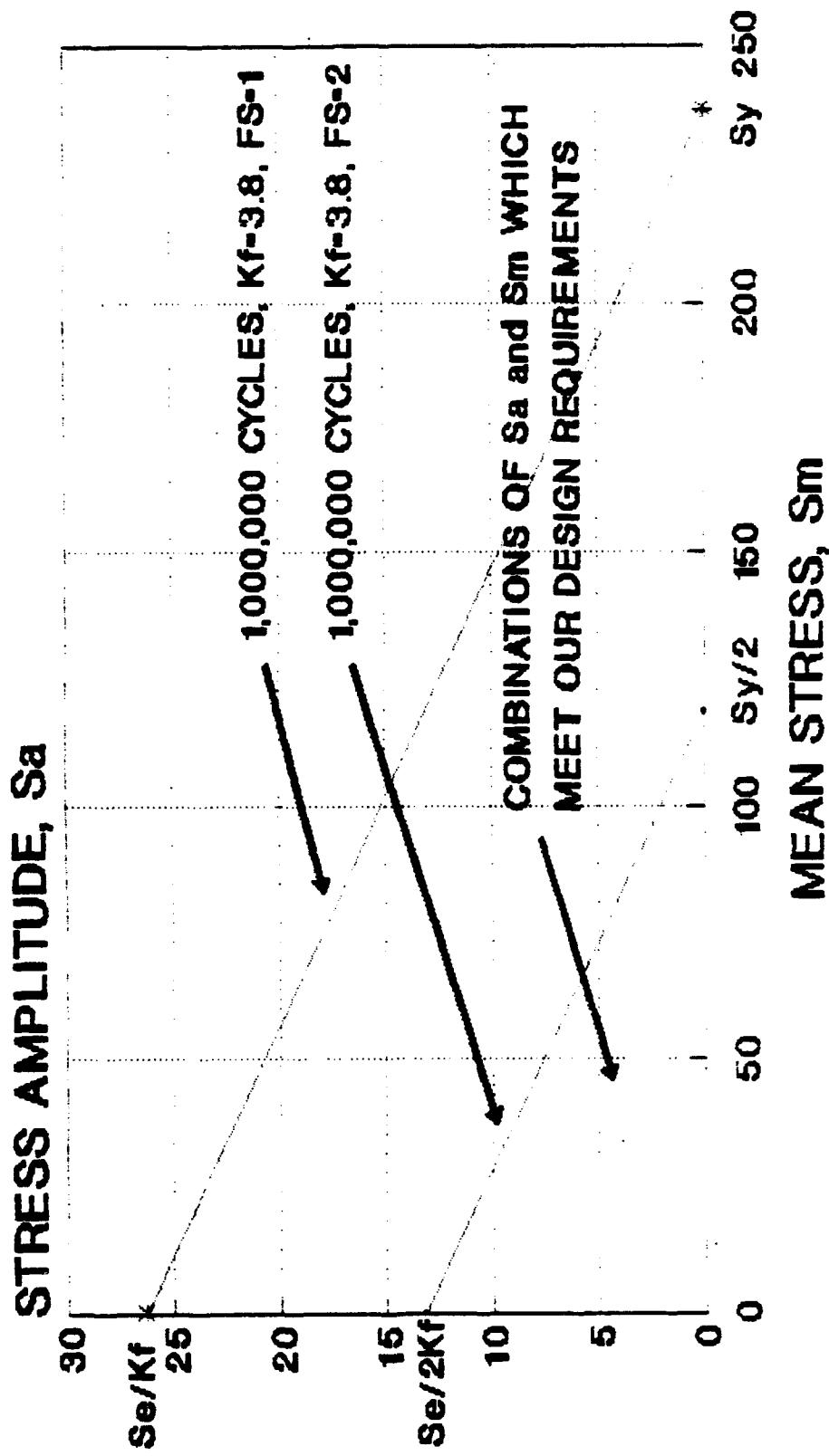


Figure 8

SODERBERG CRITERIA

ADVANTAGE OF PRELOADING

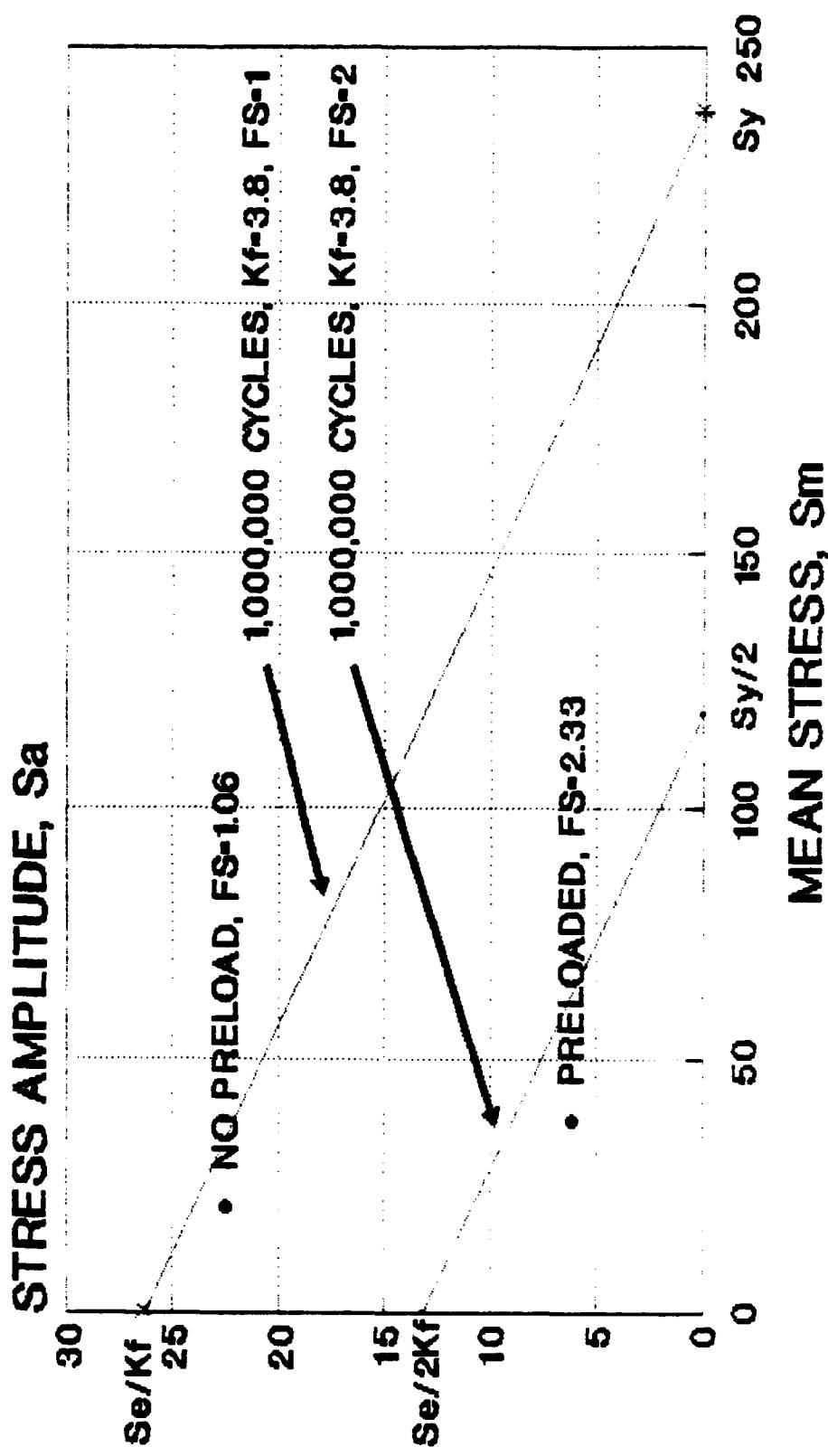


Figure 9

FATIGUE LIFE COMPARISON
PRELOADED POST vs NON-PRELOADED POST

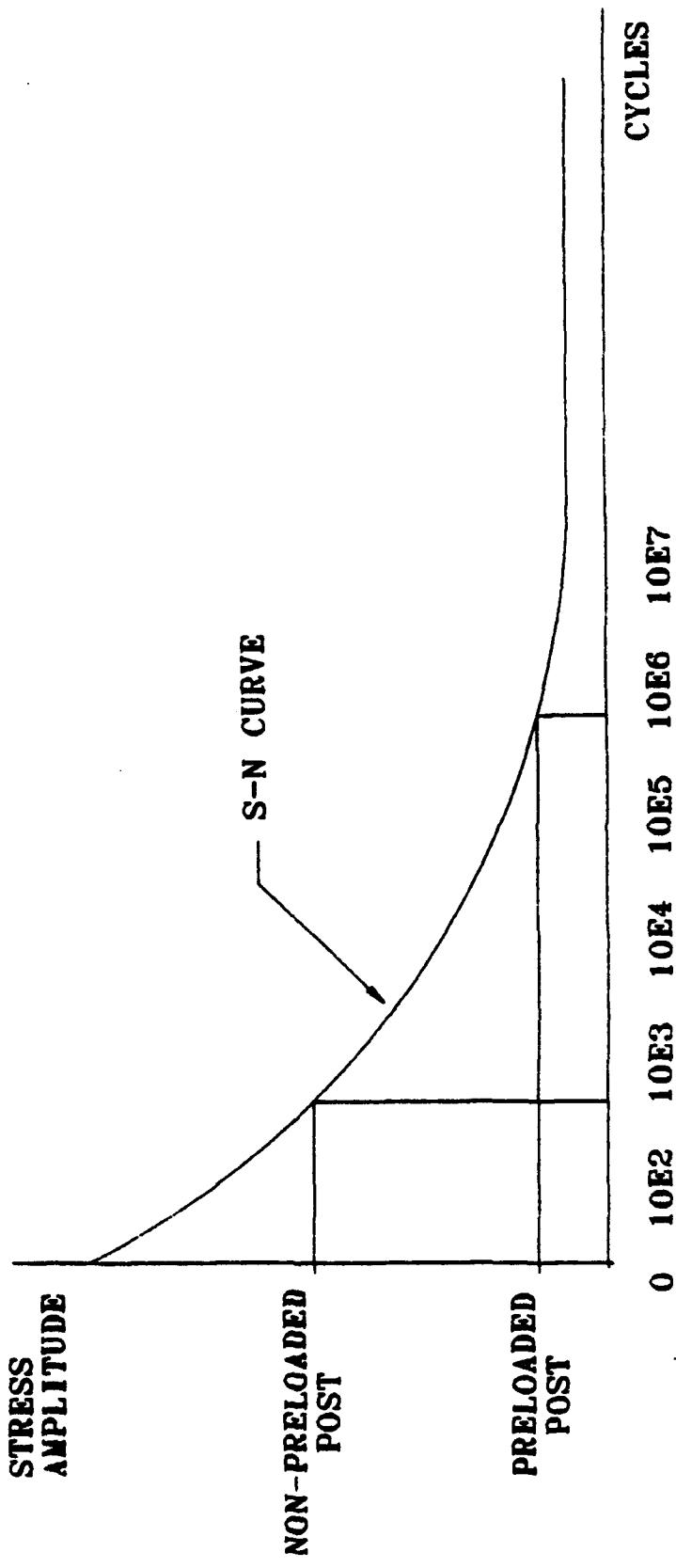


Figure 10

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